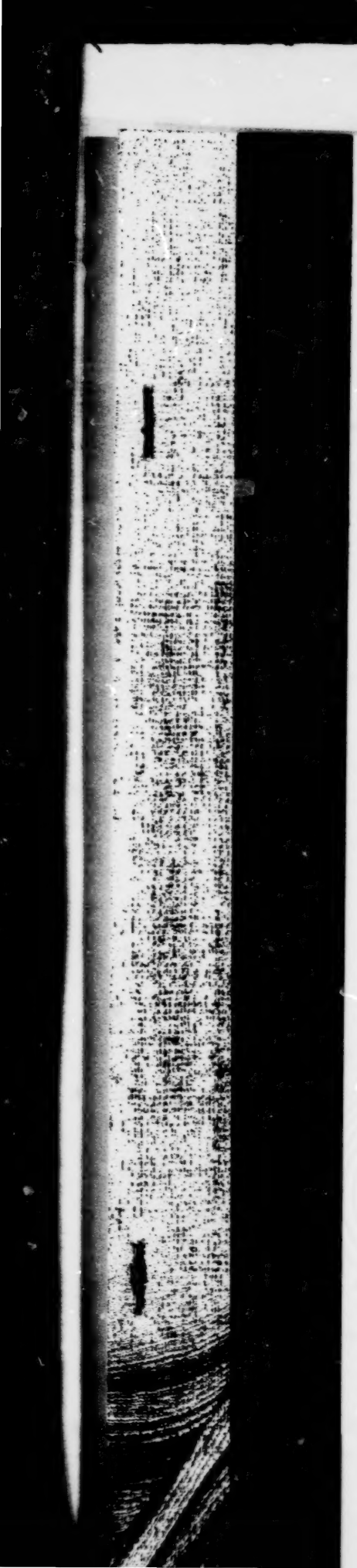

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Rapport Technique

1986/01

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A COMPARISON OF VENTILATION STRATEGIES FOR R-2000 HOUSES

J.W. Linton

Division of
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A COMPARISON OF VENTILATION STRATEGIES FOR R-2000 HOUSES

ÉTUDE COMPARATIVE DE TECHNIQUES DE VENTILATION POUR MAISON R-2000

J.W. Linton

Technical Report

1986/01

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ABSTRACT

This report examines three balanced air flow ventilation devices (exhaust and supply fans (E&SF), air-to-air heat exchanger (ATAHE) and exhaust air heat pump (EAHP)) which can be used to increase the ventilation rate in a R-2000 (tightly constructed) type house.

Two sizes of the EAHP were examined with and without inlet air electric heaters. The EAHP's and the ATAHE were each compared to the E&SF's to determine the most economical choice of device for a range of Canadian climates. Based on current energy prices, the EAHP's and ATAHE were both found to be economically attractive, except in the case where space heating is provided by low cost gas; in which case E&SF's are indicated. It was found that an EAHP could supply a substantial portion of the annual heating requirements for an energy efficient R-2000 type house.

RÉSUMÉ

Dans ce rapport on examine trois dispositifs à écoulement d'air équilibré (les ventilateurs aspirants ou soufflants (E&SF), l'échangeur de chaleur air-air (ATAHE) et la pompe à chaleur à l'air usé (EAHP) lesquels peuvent être utilisés afin d'augmenter l'efficacité de ventilation à type R-2000 (construction étanche) de la maison.

Deux EAHP, de dimensions différentes ont été examinés avec et sans chauffage électrique à l'entrée d'air. L'EAHP et l'ATAHE ont été comparés, l'un après l'autre avec les E&SF, afin de déterminer l'appareil le plus économique pour la gamme des climats canadiens.

En se fondant sur les prix actuels de l'énergie, on a trouvé que l'EAHP et l'ATAHE sont économiquement attractifs sauf dans le cas du chauffage au gaz bon marché; en ce cas les E&SF sont préférables. On s'est aperçu que l'EAHP peut fournir une partie importante du chauffage annuel dans une maison R-2000 bien isolée thermiquement.

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A COMPARISON OF VENTILATION STRATEGIES FOR R-2000 HOUSES

1.0 INTRODUCTION

In the past, Canadians have relied on the natural infiltration of outside air through cracks and other openings to ventilate our homes during the heating season. Since the early 1970's, and the advent of steeply rising energy costs, houses have been built to substantially reduce air leakage and the home heating bill. A new standard for Canadian housing has been set by the Federal Government's R-2000 Super Energy Efficient Housing Program.

Although significant energy savings have been realized, the reduction in the rate of natural ventilation has created a concern for air quality. It has been found that if the ventilation rate of a house is lowered to below 0.5 of an air change per hour (ACH), problems with indoor air quality can arise (Scheuneman, 1982 and Tenwolde, et al., 1984). With such a low rate of fresh outdoor air entering the house (and corresponding low rate of 'stale' air leaving the house) pollutants and moisture can build up to what may be unacceptable levels.

The need for controlled ventilation in the home, together with the need for energy conservation has focused attention on heat recovery from exhaust air.

This report looks at three different mechanical devices which could be used to increase the ventilation rate of tightly built R-2000 type houses. The three devices are an exhaust and supply fan (E&SF), air-to-air heat exchanger (ATAHE), and an exhaust air heat pump (EAHP). E&SF's and ATAHE's for residential applications are relatively straight forward devices with many commercial examples available on the market. However, the EAHP is a new concept which is currently very much in the development stage. Therefore this report will look at some alternative designs for the EAHP.

2.0 VENTILATION STATEGIES

The three systems compared are exhaust and supply fans (E&SF), an air-to-air heat exchanger (ATAHE), and an exhaust air heat pump (EAHP). All three ventilation devices would be classified as balanced, where the airflow forced out of the house equals the air flow rate drawn into the house by the device, and hence the pressure inside the house is affected very little. For very tightly constructed houses with natural infiltration rates lower than say 0.2 ACH, an exhaust only ventilation system is not considered appropriate because of the very low pressure needed in the house to encourage sufficient air to leak through the 'tight' building shell. This low pressure could cause problems with the building such as in opening exterior doors and windows (Shaw, 1983).

To allow comparison of the three devices, a house with a natural infiltration rate of 0.2 ACH was considered. (This infiltration rate would be typical of tightly constructed R-2000 houses). This detached, split entry style house with a heated basement was considered typical of the new R-2000 type construction. The house is designed to meet the average R-2000 construction specifications based on details provided in an R-2000 Home Monitoring Report by the National Research Council Canada (Dumont, 1984), and is described in Appendix A. The ventilation devices were sized to produce a total ventilation rate of 0.5 ACH in this house. This ventilation rate is considered to represent an acceptable standard for maintaining air quality (Scheuneman, 1982 and Tenwolde, et al., 1984).

Shaw (1983) has presented findings on the effect of balanced ventilation systems on the ventilation of a typical, tightly-constructed two-storey detached house with basement. Exhaust was from the basement and supply was to the return duct of the central, forced-air heating system. The air change rate was measured using the tracer gas (N_2O)concentration decay method. Tracer gas was injected into the furnace duct system and air samples for measuring the tracer gas concentration were also taken from the duct system. The furnace fan was run continuously during these tests and tests were run during a heating season with a range of indoor to outdoor temperature differences, wind speeds and wind directions.

During the colder winter months (outdoor temperatures below -10°C) the natural air infiltration rate for the test house was found to vary very little with temperature difference and wind speed or direction. An average value of 0.26 air changes per hour (ACH) was found for this test house.

For the balanced systems, Shaw found that during the colder winter months the ventilation rate for the house, I, was not just the sum of the natural infiltration rate, I_o, and forced ventilation rate V_v, but is a linear relationship;

$$I = I_o + RV_v \qquad (1)$$

where the proportionality constant, R, is given by

$$R = 0.13\ (\Delta t)^{1/2} \qquad (2)$$

and Δt is the temperature difference between inside and outside. The effect of wind on the ventilation of the house was much less than the temperature difference effect and was ignored in this correlation.

From Table 1 it is apparent that to ensure adequate ventilation over the whole heating season, a forced ventilation rate of 0.5 ACH is required for the balanced system.

This apparent anomaly is mainly due to the fact that the various volumetric flow rates were not corrected for temperature. Therefore, as the ambient air increases in temperature, the fresh air entering the house becomes less dense and so higher volumetric flow rates are required to achieve the same total ventilation rate of 0.5 ACH. Also, at indoor to outdoor temperature differences below 30°C the natural ventilation rate will probably decrease also, thereby requiring an increase in the mechanical ventilation rate to compensate.

The ventilation devices will be assumed to operate with an exhaust and supply rate, V_v, of 70.8 std L/s corresponding to 0.5 ACH. This ventilation rate will be maintained for the entire heating season of nine months (6552 hours) to ensure adequate ventilation. The compressor of the EAHP will cycle on and off with the demand from the first stage of the house thermostat. The second stage of the thermostat would be connected to the central furnace.

Staff at Lawrence Berkeley Laboratory determined an average air-to-air heat exchanger effectiveness of about 0.6 for two 'good' quality ATAHE's (Fisk, et al., 1984). It will be assumed that this figure includes any internal cross stream leakage and allows for losses due to defrosting the device.

The characteristics of the three ventilation systems compared are illustrated diagramatically in Figure 1.

3.0 DESIGN OF EXHAUST AIR HEAT PUMP (EAHP)

For heat pumps, the heat flow rate from the condenser, Qc, the heat flow rate to the evaporator, Qe, and power consumed by the compressor, Pc, are related in the following way (ignoring the small losses present).

$$Qc = Qe+Pc \tag{3}$$

A coefficient of performance (COP) can be defined as in (4) (not allowing for the power to operate fans).

$$COP = \frac{Qc}{Pc} \tag{4}$$

$$\text{Therefore, } Qe = Qc \left(1-\frac{1}{COP}\right) \tag{5}$$

For a constant condenser heat delivery, Qc, it is apparent from (5) that the heat recovery by the evaporator, Qe, will increase as the COP increases and the purchased power, Pc, will decrease.

Conversely, for a system with a fixed evaporating temperature, the COP and energy recovery increases with a falling condenser temperature.

In the case of the EAHP considered, the evaporator is working with a fixed flow rate of exhaust air for a particular size house (70.8 std L/s (150 scfm) for 0.5 air changes per hour in a 212 m^2 house), because the EAHP is a balanced ventilating device, the condenser is also working with a fixed flow rate of supply air (70.8 std L/s (150 scfm)). The evaporator section of the EAHP would operate with a constant inlet temperature of 21.6°C (71°F), while the condenser section which uses fresh air from outdoors would operate in a temperature range of -40°C to 15°C (-40°F to 59°F). When designing the EAHP the two extremes of operating temperature had to be considered. The low temperature design condition is based on CSA Standard C439-M1985, Standard Methods of Test for Rating the Performance of Heat Recovery Ventilators, which requires the ventilation unit to operate continuously for 72 hours at a fresh air supply temperature of -25°C (-13°F) and relative humidity of 50% to 100%. The high temperature design condition is 15.0°C (59°F), above this point no heating of the inlet air stream is required, see Table A1.

Operation with very low air supply temperatures would result in low condensing temperatures. These low temperatures can lead to overloading of the compressor motor and erratic throttling valve operation.

To prevent the operating problems encountered at low condensing temperatures a heat pipe will be installed between the exhaust and supply air streams, see Figure 2 for the equipment arrangement and typical operating conditions. The heat pipe will also allow a smaller refrigeration system to be used to recover the maximum heat from the outlet air stream.

The effectiveness of the heat pipe was selected at 30%. This would allow for maximum energy recovery from the exhaust air, but not drop the exhaust air temperature to a point where frosting could occur on the heat pipe, and would maintain an air temperature high enough to defrost the heat pump evaporator if it became frosted.

A problem associated with an evaporating temperature below -1°C is the frosting of the heat pump evaporator. This means that the heat pump compressor will have to be turned off periodically and the evaporator defrosted. Typically, the defrost would be accomplished by using warm exhaust air.

A conventional, vapour compression heat pump operates on a reverse Rankine cycle and for a particular system with a fixed condensing temperature, the COP increases almost linearly with evaporating temperature. Therefore, the higher the evaporating temperature, the greater the energy recovery of the heat pump.

A report by Hawken (1984), which compares frosting and non-frosting exhaust air heat recovery heat pumps, and actual field tests of a prototype frosting exhaust air heat recovery heat pump (Blackmore, 1984) designed by Hooper and Angus (1982) indicates that a smaller system with non-frosting evaporator will recover similar amounts of energy from the exhaust stream as the larger frosting evaporator type. Based on the above reports a non-frosting evaporator was selected.

The sizing of the evaporator, condenser and compressor is based on the following criteria:

- 70.8 std L/s (150 scfm) air flow for evaporator and condenser
- 21.6°C (71°F) indoor air at 40% R.H. (exhaust)
- -25°C to 15°C (-13°F to 59°F) outdoor air at 50% to 100% R.H. (supply)
- 30% heat pipe effectiveness
- Non-frosting evaporator (at -25°C supply air)

The individual components were selected and the system performance evaluated using a heat pump system design method (Hawken, et al., 1984) which has proven to be reliable in predicting the performance of real heat pump systems.

Two sizes of EAHP were evaluated, a small EAHP will have an evaporator leaving air temperature just above freezing to avoid frosting of the evaporator when outdoor air is down to -25°C (-13°F). The large version of the EAHP will remove additional energy from the exhaust air but would frost the evaporator at low outdoor temperatures. To avoid this an evaporator pressure regulator will be installed on the suction line to the compressor to prevent the evaporator from reaching a temperature that would cause freezing-up of the coil.

Also evaluated is the addition of a 0.75 kW plus 1.5 kW staged electric heater in the small EAHP and a 1.0 kW heater in the large EAHP. The heaters would be located on the incoming air side between the heat pipe and the condenser; see Figure 2 for component layout. The electric heaters were sized to maintain the supply air to the house above 20°C (70°F).

The electric heaters provide the following benefits:

- At low outdoor air temperature the fresh air supplied to the house will remain above 20°C (70°F) which would eliminate special supply ducting arrangements to avoid drafts in the house. If the electric heaters are not used at low outdoor temperatures the inlet temperatures can reach uncomfortably low supply air temperatures, which require locating supply air ducts where this cool air will not cause a draft (see Installers Manual for Heat Recovery Ventilators, Heating, Refrigerating and Air Conditioning Institute of Canada).

- The heater would raise the temperature of inlet air to the condenser which in turn would raise the suction temperature and prevent evaporator frosting

The electric heaters would become mandatory in climates with extended periods of low ambient temperature below -25°C (-13°F).

Table 2 includes a description of the design specifications of the large and small EAHP's main components.

Figures 3 to 10 show that the capacity of the EAHP's actually increases with the lowering of the outdoor temperature. The COP of the EAHP's also increases with lowering outdoor temperature, except for the EAHP's with electric heaters which experience a large drop in COP as the heaters are turned on, see Figures 6 and 10.

From Figure 11 which is a typical psychrometric chart of the small EAHP with heaters (also refer to Figure 2) the exhaust air conditions from the EAHP are represented by the line with points 1, 2 and 3. The indoor air entering the EAHP is at the condition of point 1, as the air passes through the heat pipe it is cooled and there is a slight moisture loss, it then exits the heat pipe at the condition of point 2. The exhaust air then passes through the evaporator of the EAHP where it is cooled and moisture is condensed to the condition of point 3, which is discharged outdoors. The supply air conditions are represented by the line with points 4, 5, 6 and 7. The outdoor air entering the EAHP is at the condition of point 4 (-25°C, 30% to 100% RH), as the air passes through the heat pipe it is heated to the condition of point 5 (-10.9°C), it then exits the heat pipe and it is further heated to the condition of point 6 by the 1.5 kW electric heater. The supply air is then heated to the condition of point 7 by the condenser, and supplied to the house at this condition.

Static pressure gains required for the fans were calculated from curves supplied by a heat exchanger manufacturer (OEM Coil Manual). An allowance was also included for cabinet and external duct losses. The supply and exhaust fan of each ventilation device would be identical, the supply and return duct work would require balancing dampers to ensure equal air flows. Table 3 summarizes the fan requirements for all the devices considered in the report.

4.0 COMPARISON OF VENTILATION DEVICES

4.1 Basis of comparison .

For this comparison, the energy required to heat and ventilate the house described in Appendix A was determined for three cities in Canada. Vancouver, Montreal, and Saskatoon were chosen to represent the range of climates experienced in this country with heating degree (Celsius) days (18°C base) of 3007, 4471 and 6077 respectively.

In each location, six different heating and ventilating strategies were examined. In all cases, the house relied on a gas or electric furnace as the primary heating device. For the base case, ventilation of the house was accomplished with exhaust and supply fans (E&SF). In the five other cases, the exhaust and supply fans were replaced by an exhaust air heat recovery device; either an air-to-air heat exchanger (ATAHE) or a small exhaust air heat pump (EAHP) with and without electric heaters or a large exhaust air heat pump (EAHP) with and without electric heater.

The costs of various fuels on a regional basis were supplied by the Energy Statistics Handbook from Energy, Mines and Resources Canada. Oil heating is not considered as an alternative here. Table 4 shows the cost of energy produced and includes an estimate for typical seasonal furnace efficiency values. As the new generation of high efficiency gas furnaces increases in its market share, the cost of energy from gas will decrease slightly.

In order to simplify heat loss calculations the heat loss/gain for the base house with exhaust and supply fans is assumed to have a constant 0.7 ACH (no allowance for outside temperature differences). The 0.7 ACH includes 0.2 ACH of natural infiltration plus 0.5 ACH for forced ventilation. The house with the ATAHE (60% effectiveness) is assumed to have 0.4 ACH. The 0.4 ACH includes 0.2 ACH of natural infiltration plus 0.5 ACH for forced ventilation with 60% of the exhaust heat recovery, leaving 0.2 ACH to be heated to indoor temperature. The heat loss/gain for the base house and ATAHE house are given in Table A1.

From Table A1 it is apparent that there is not a net heat loss at outdoor temperatures above 15.0°C (59°F) and therefore the energy loss due to ventilation is only for outdoor temperatures of 15.0°C (59°F) and below.

The seasonal heating requirements for the base house and ATAHE house are given in Tables A2 and A3.

For the ATAHE a constant energy recovery of 60% of the heat loss is assumed (Fisk, et al., 1984). All the ventilation devices are assumed to run continuously for 6552 hours. With the EAHP's, the evaporator and condenser fans run continuously and the compressor will operate on demand from the house thermostat.

The results of the analysis are summarized in Tables 5 and 6. Appendix B details how the various values were calculated. Estimates of the costs of the devices are given in Table 7. The costs for the ventilation devices were partly based on the Hooper and Angus report (1982). The installation of the ventilation devices is based on recommendations in the Heating, Refrigerating and Air Conditioning Institute of Canada Installer's Manual for Heat Recovery Ventilators (1985).

The simple payback (SPB) ratio of initial cost divided by the cost savings per year was used for the economic analysis. The use of the SPB is easy to apply and is generally understood by the public. Also, it is considered that unless a non-essential system can show a simple payback period of less than about five years, based on current costs, then it is not economically attractive for the home-owner to purchase.

4.2 Base Case Versus ATAHE and Small and Large EAHP

Under the column heading "Energy Delivered by Ventilation Device" in Table 5 it shows that the small and large EAHP's save considerably more energy than the ATAHE in all cases considered. As expected, the ATAHE recovers a fairly constant percentage of the energy lost by the exhaust air, with a slightly downward trend in warmer climates as the fan energy has more influence. Conversely, Table 8 indicates the EAHP saves more energy as the climate gets colder, as can seen by comparison of the heating required by the house for each city compared to the useful heat supply by the EAHP's. This is due to the increased running time of the compressor. The capacity of the EAHP actually increases as the outdoor temperature drops. This is due to the lowering of condensing temperature which will decrease the power requirement of the

compressor and hence raise the COP of the heat pump; see Figures 3 through 10 for capacity and COP vs. outdoor temperature. These figures indicate that the overall system COP including heat pipe but excluding fans is a high as 6.2 at -25°C (-13°F) for the small EAHP without heaters. Figures 5 and 6 illustrate the performance of the small EAHP with electric heaters. These figures show the effect on overall COP and capacity as the electrical heaters are turned on by a drop in the supply air temperature. The effect of the electric heaters is similar for the small EAHP which has two heater stages, [Figures 5 and 6] and the large EAHP which has one stage, [Figures 9 and 10].

Looking at Table 6, the economic picture is not as straight forward as the energy savings. Because of the low cost of natural gas in Vancouver and Saskatoon (slightly over 2.0 times less expensive than electricity, see Table 4), the use of EAHP is of questionable economic value in these areas, especially when electric heaters are included in the EAHP's. The ATAHE would also be of marginal economic benefit when installed in a gas heated house in Vancouver or Saskatoon.

In Montreal, where gas and electric energy prices are almost equal, the ATAHE and EAHP's can be justified economically, with payback periods ranging from 2.6 to 3.2 years for gas heating.

In the cases where electricity is the space heating source being considered, (a survey of existing R-2000 houses indicates that 63.5% are all electric heated (Dumont, 1984)) then the ATAHE and the EAHP's become very viable alternatives in all three locations, with paybacks ranging from 1.5 to 3.4 years.

The EAHP's also satisfy a significant proportion of the seasonal heating load for the R-2000 house considered (see Table 8), especially in milder climates such as Vancouver where 100% of the heating load can be met by the large EAHP. This means that in many locations, the EAHP could provide the base line heating for the house, with electrical resistance, gas or some other form of energy providing supplementary heat.

The above analysis has a weakness in that it assumes a constant temperature and relative humidity within the house. It would be expected that the humidity and, to a lesser extent, the temperature within the house will vary during the heating season, e.g. higher humidity in the warmer months and lower humidity in the colder months. This variation in humidity will affect the capacity of the EAHP. Should the humidity go too low, it is possible that the EAHP will begin to frost up and hence a control should be incorporated to detect this condition. The only way of determining the 'true' performance of an EAHP is to build several prototypes and to test them in several realistic (i.e. occupied) residential situations.

5.0 THE EAHP AS A DEHUMIDIFIER

By including some modifications to the supply and return air ducts, and adding a humidistat the EAHP could be used as a basement dehumidifier in the summer months when mechanical ventilation is not required, see Figure 12 for dehumidifier layout.

Many Canadian houses require dehumidification of the basement during the summer months to prevent moisture problems. For design conditions based on CSA Standard C22.2 No. 92-1971 for dehumidifiers, the small EAHP would remove 9.0 litres per day and the large EAHP would remove 13.3 litres per day of water from the basement of the house.

Currently, mass produced room size dehumidifiers of similar capacity retail for about $300.

The additional duct work and controls could be expected to add about $70. to the installation cost. Therefore, the EAHP is extremely attractive in that it can pay for itself in heating costs savings in under three years and will also provide basement dehumidification in the summer; thus saving the cost of purchase of a dehumidifier.

6.0 CONCLUSIONS

An analysis of the heating and ventilating requirements of a typical, 'comfortable', detached house of the new R-2000 type construction was conducted for three representative locations in Canada. The effect of three different ventilation devices on the cost of heating and ventilating this house was examined for each of the three locations. An exhaust and supply fan (E&SF), air-to-air heat exchanger (ATAHE) and several versions of an exhaust air heat pump (EAHP) were examined. The results of the analysis are summarized below.

The ATAHE and EAHP are both economically attractive for the R-2000 type house when compared to exhaust and supply fans. The only exception to this is for a gas heated house in western Canada where the energy cost ratio between electric and gas heat is $\geq$ 2. When the EAHP was installed in electrically heated houses (which accounts for 63.5% of R-2000 houses built to date) it showed simple paybacks ranging from 1.7 to 3.4 years. These payback periods were based on current 1985 energy prices.

An EAHP can also supply a substantial portion of the seasonal house heating requirements. In milder climates, the EAHP can provide 100% of heating requirements.

The EAHP can be converted to a basement dehumidifier during summer months with a simple modification of the supply and return air ducts.

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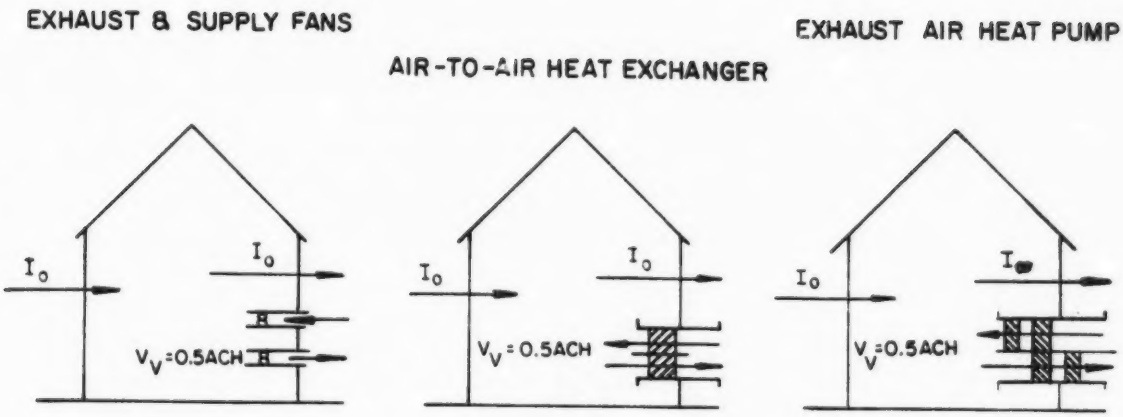
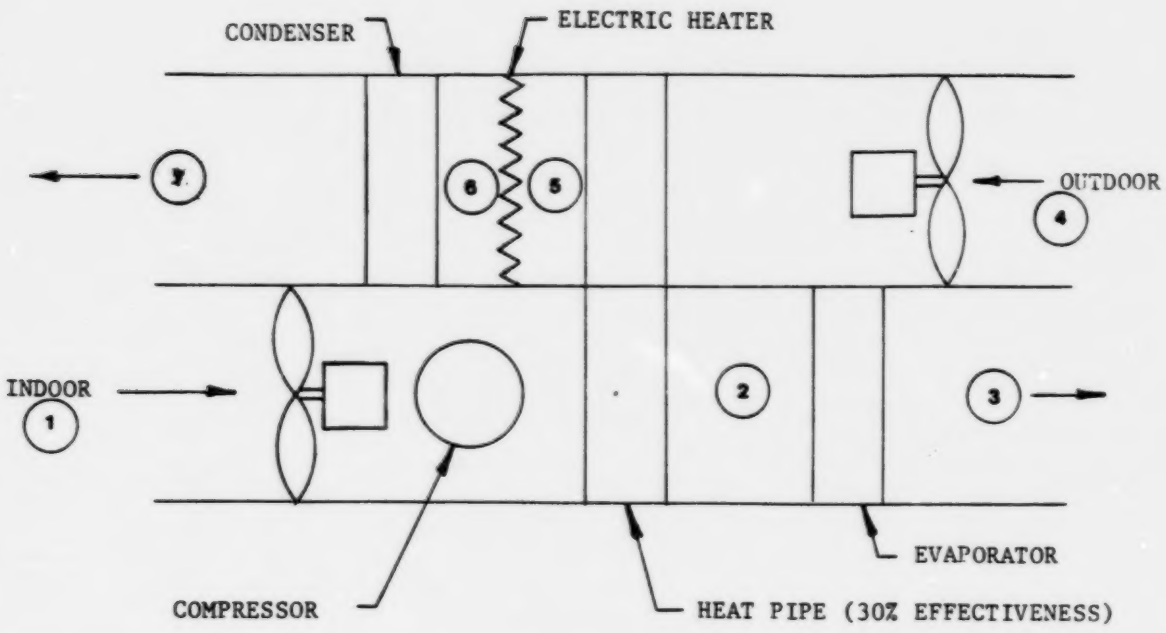


FIG. 1 SCHEMATIC OF THREE VENTILATION SYSTEMS



TYPICAL OPERATING AIR TEMPERATURES AND RELATIVE HUMIDITIES

ITEM	SMALL EAHP	SMALL EAHP WITH HEATER	LARGE EAHP	LARGE EAHP WITH HEATER
1	22 °C 40%	22 °C 40%	22 °C 40%	22 °C 40%
2	7.9 °C 90%	7.9 °C 90%	7.9 °C 90%	7.9 °C 90%
3	1.6 °C 100%	1.9 °C 100%	0.55 °C 100%	0.55 °C 100%
4	-25 °C 30% - 100%	-25 °C 30% - 100%	-25 °C 30% - 100%	-25 °C 30% - 100%
5	-10.9 °C	-10.9 °C	-10.9 °C	-10.9 °C
6	-10.9 °C	6.6 °C	-10.9 °C	0.72 °C
7	3.9 °C	21.6 °C	12.8 °C	22.9 °C

FIG. 2. EXHAUST AIR HEAT PUMP EQUIPMENT ARRANGEMENT AND TYPICAL OPERATING CONDITIONS

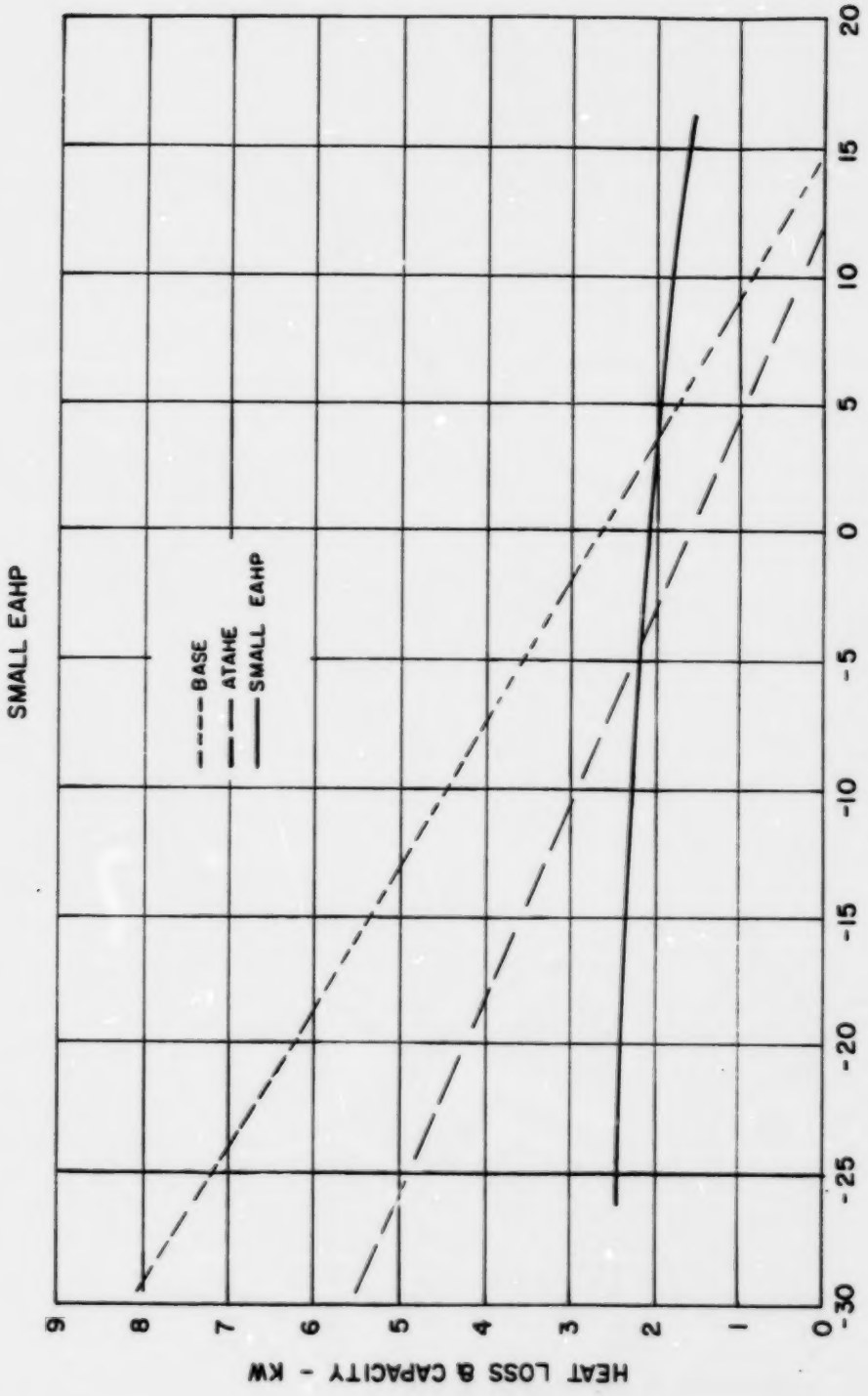


FIG. 3 HEAT LOSS & EAHP CAPACITY

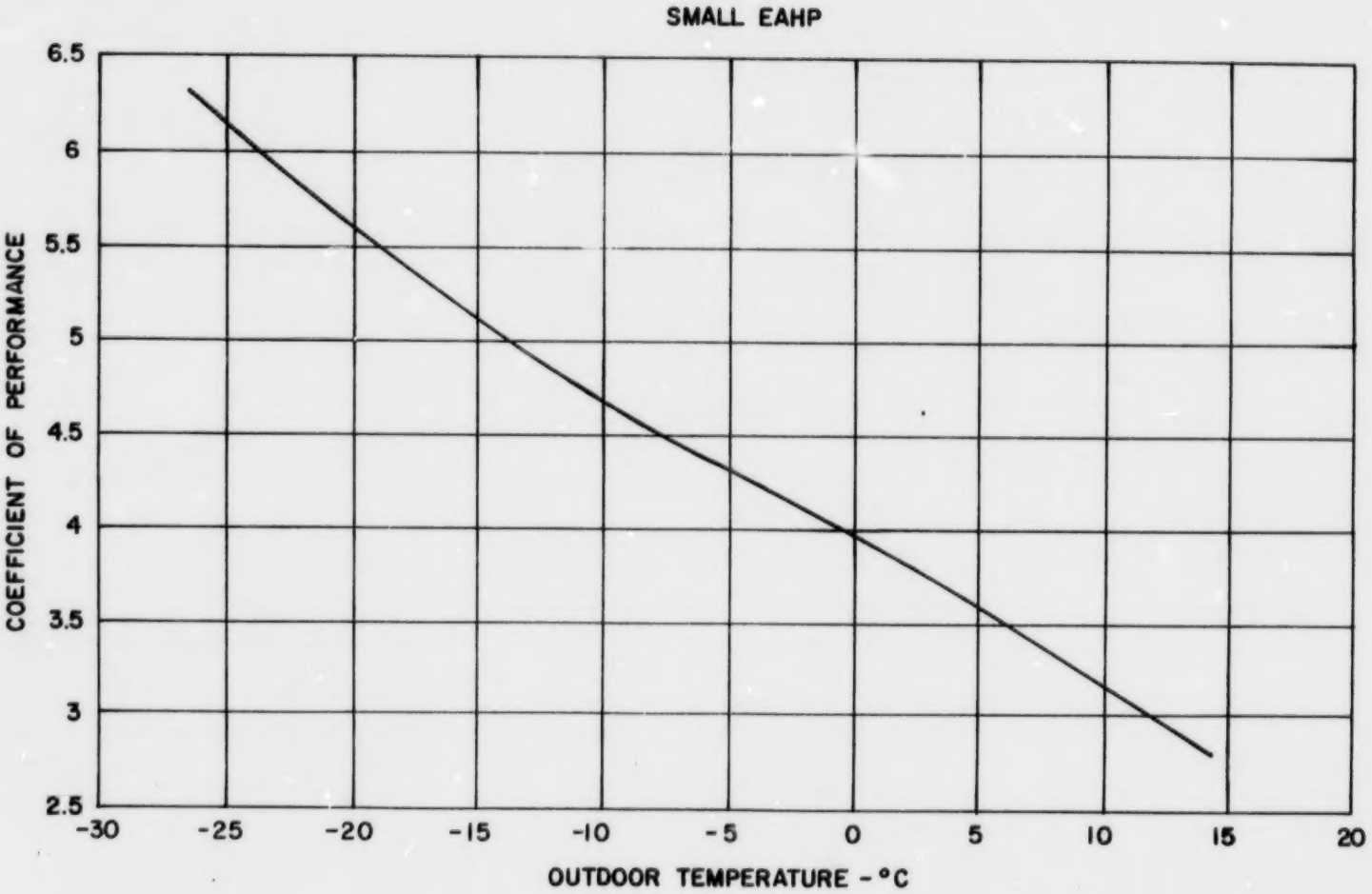


FIG. 4 COP VS OUTDOOR TEMPERATURE

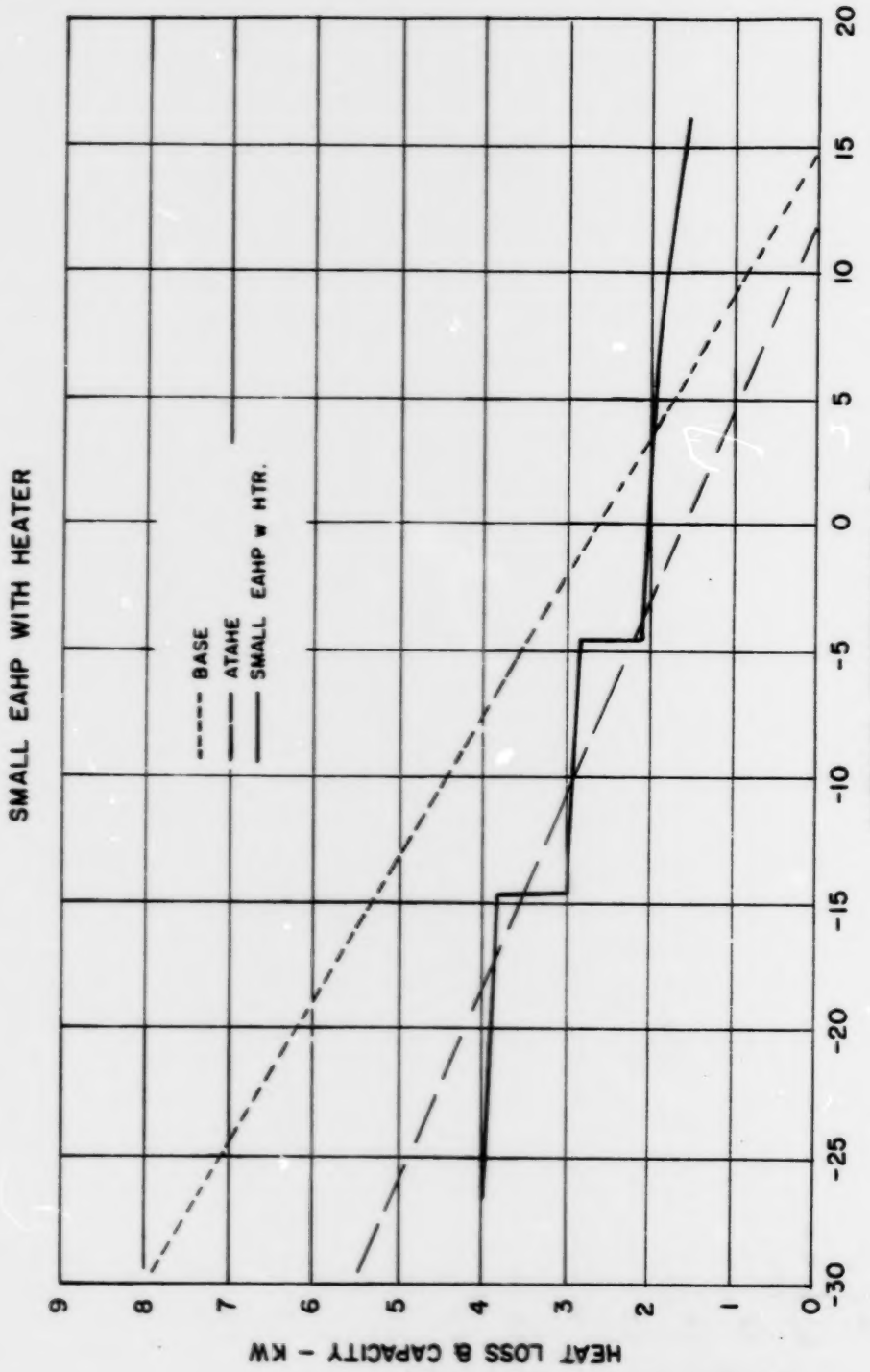


FIG. 5 HEAT LOSS & EAHP CAPACITY

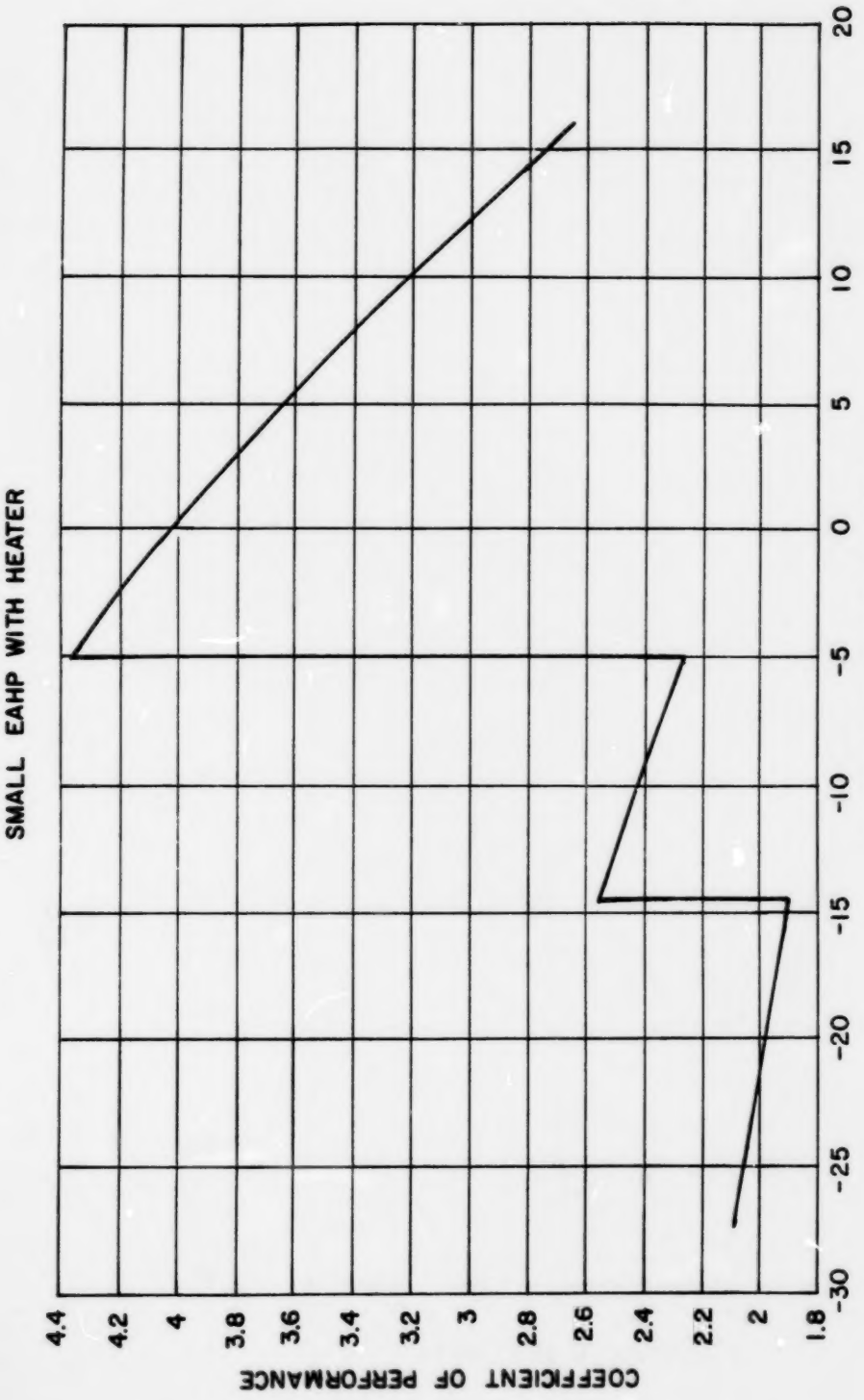


FIG. 6 COP VS OUTDOOR TEMPERATURE

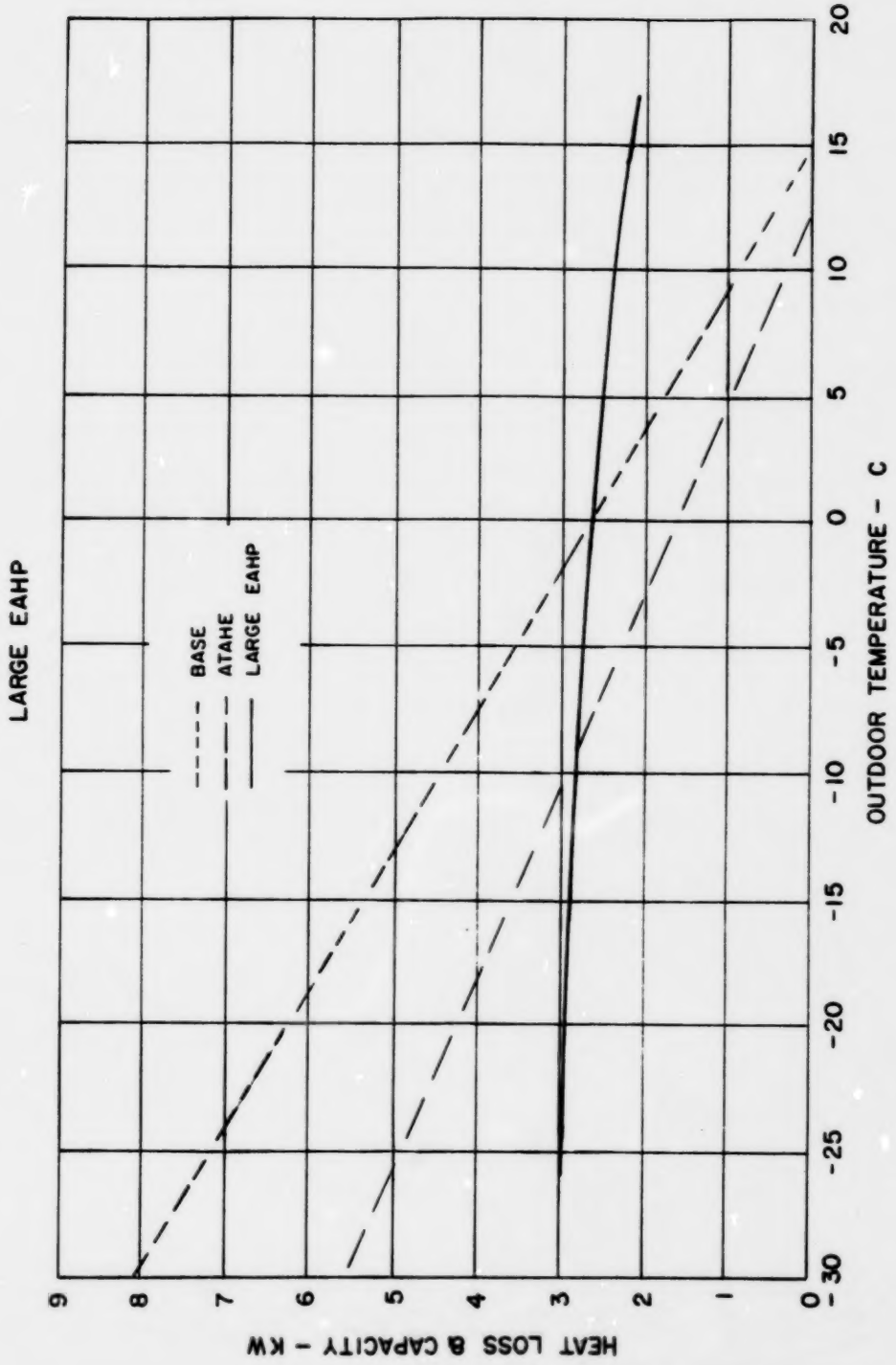


FIG. 7 HEAT LOSS & EAHP CAPACITY

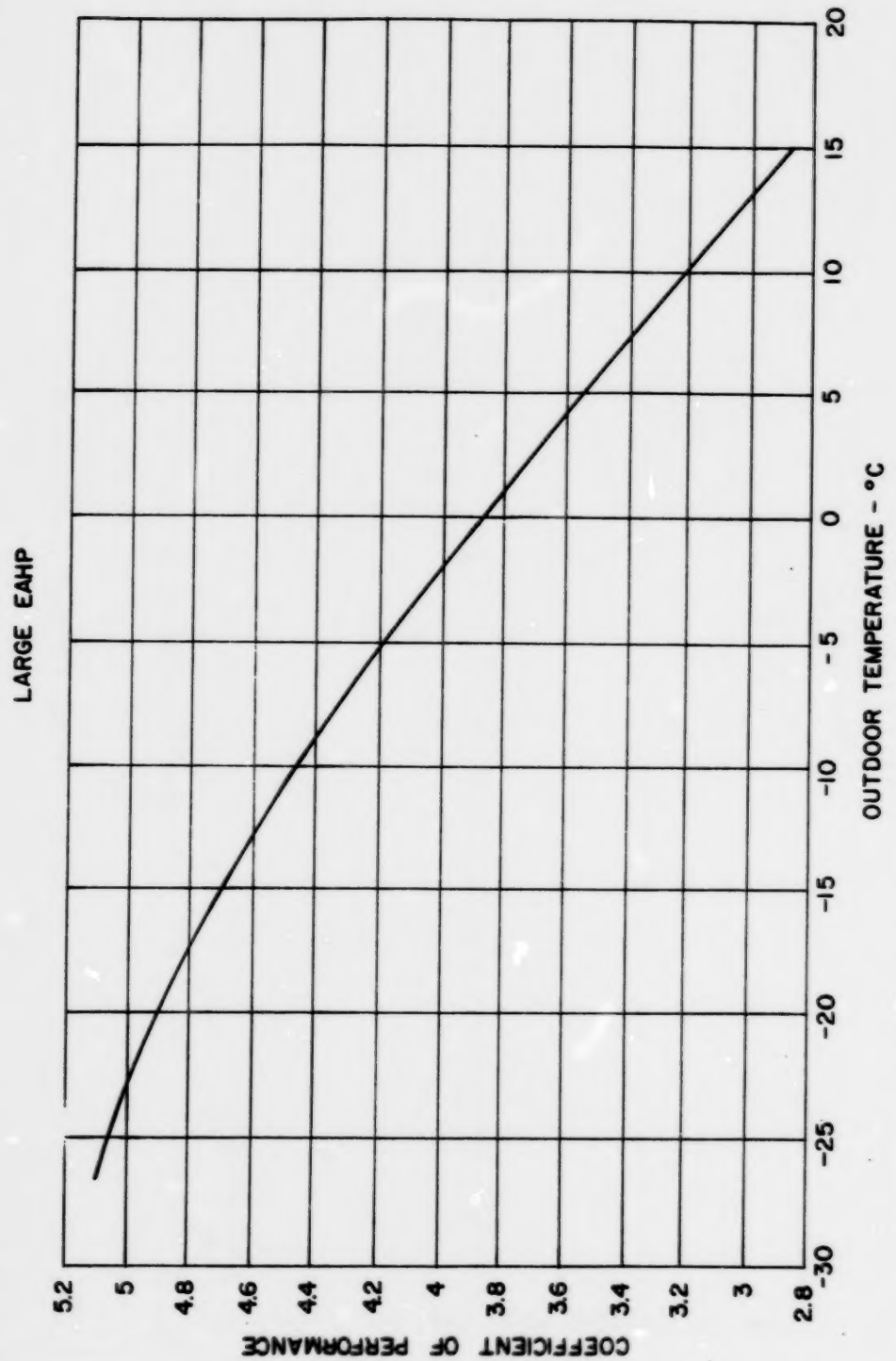


FIG. 8 COP VS OUTDOOR TEMPERATURE

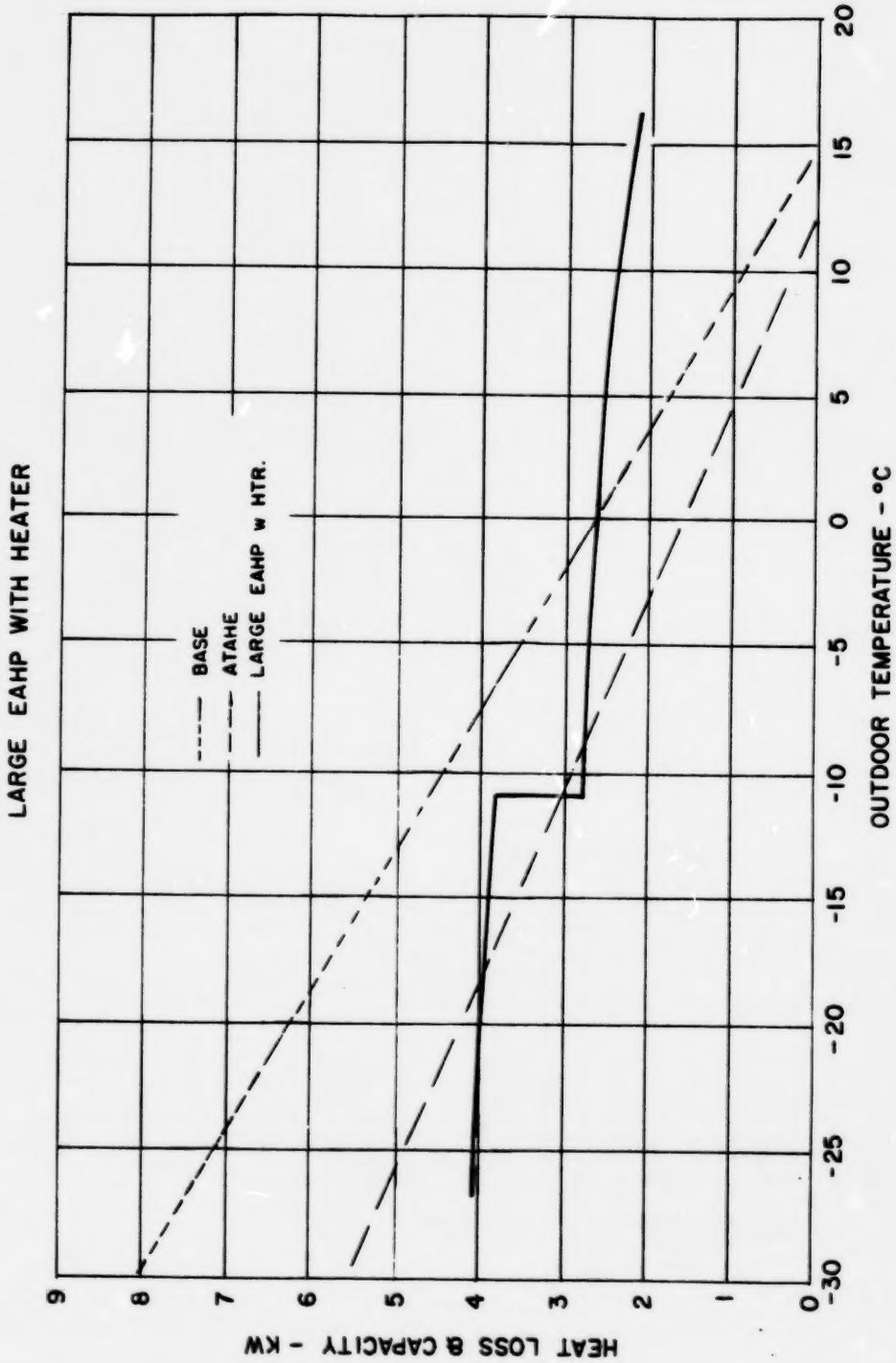


FIG. 9 HEAT LOSS & EAHP CAPACITY

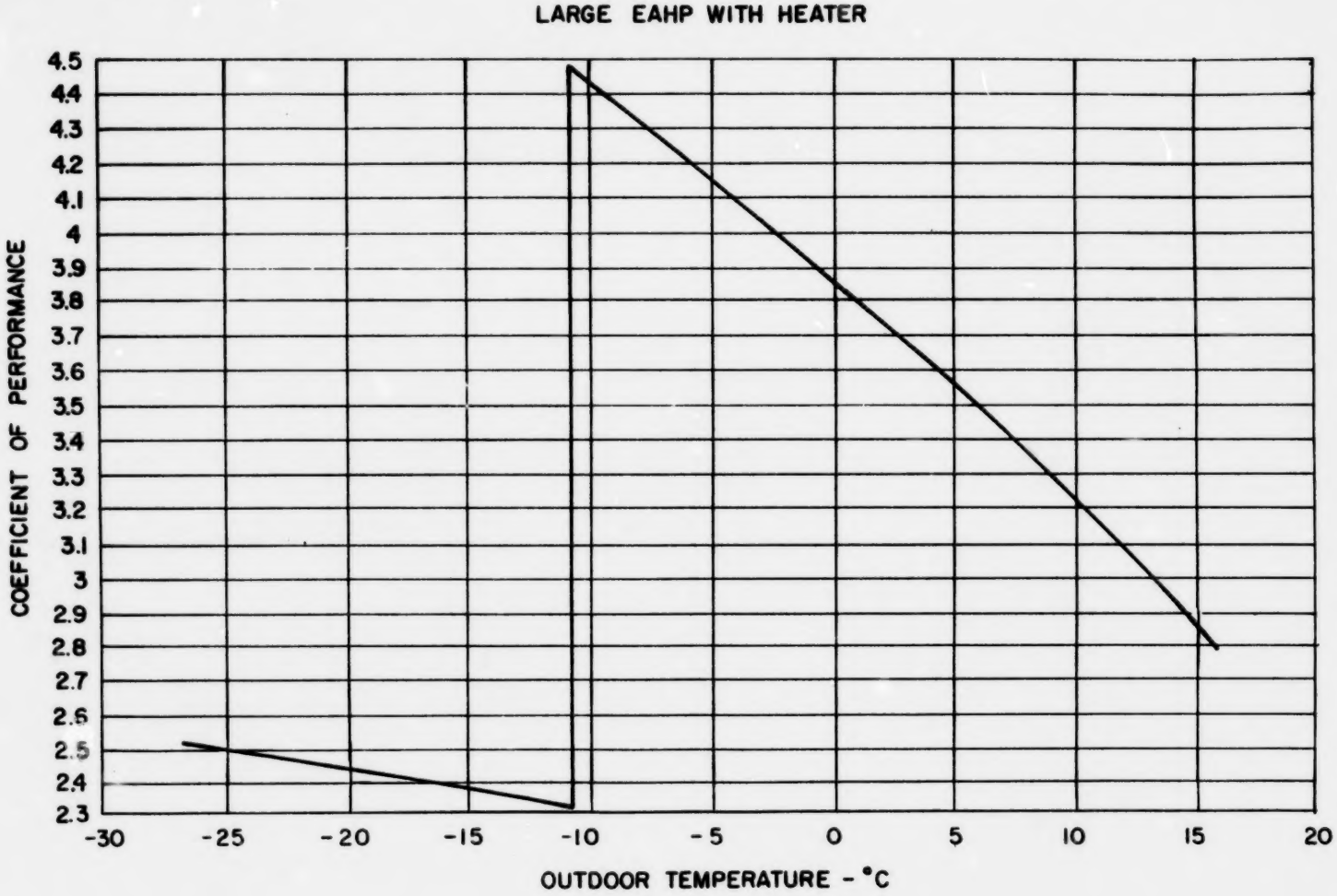


FIG. 10 COP VS OUTDOOR TEMPERATURE

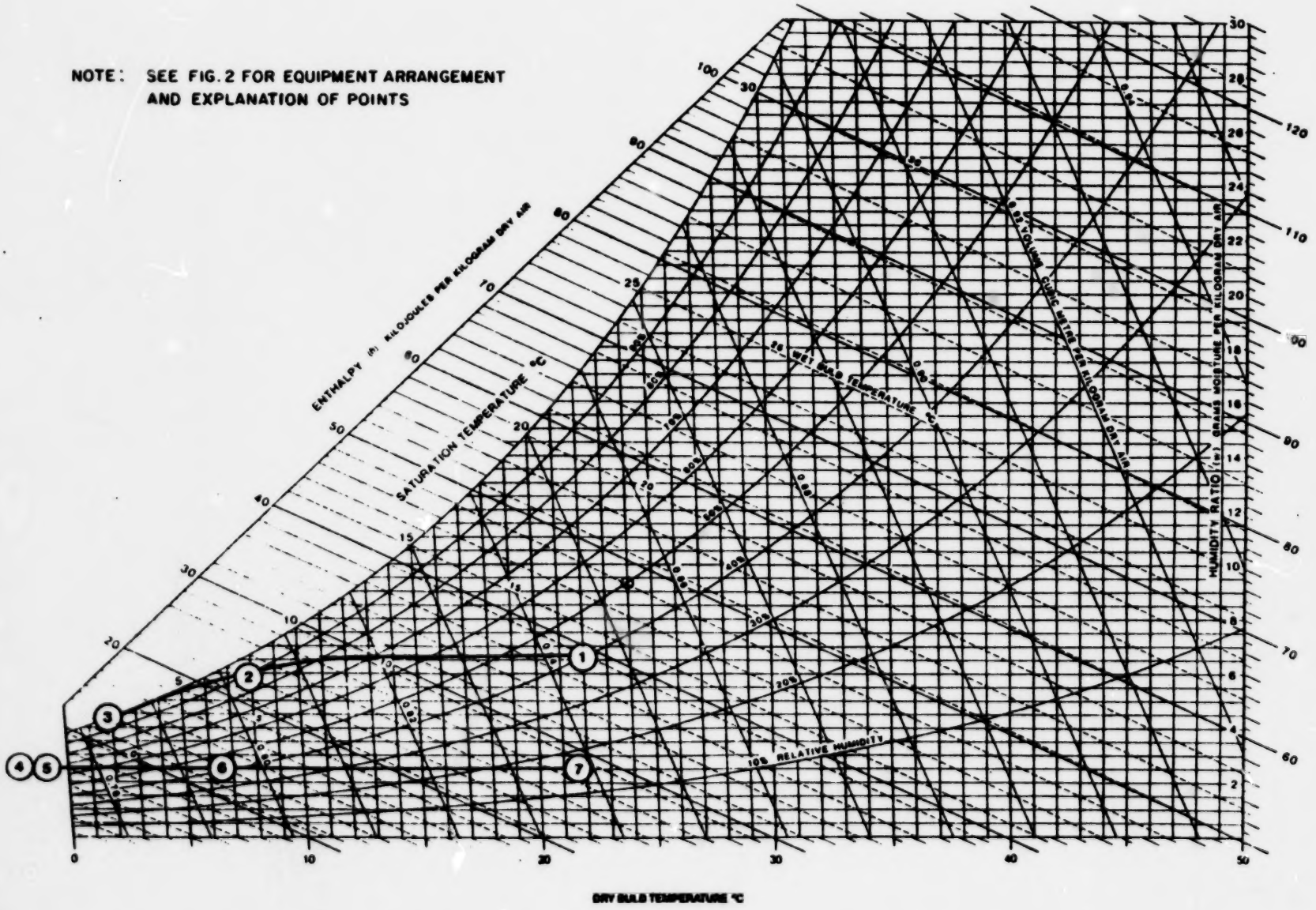
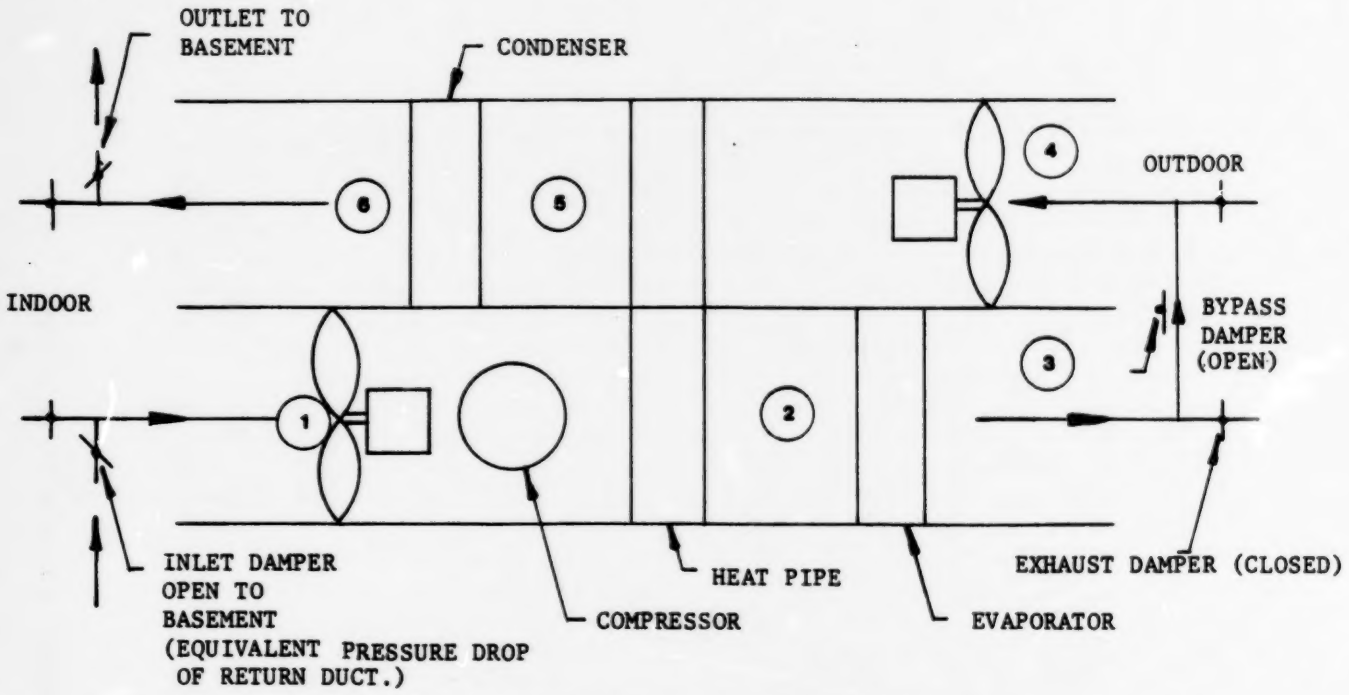


FIG. 11 TYPICAL PSYCHROMETRIC CHART OF SMALL EAHP WITH HEATERS



TYPICAL OPERATING AIR TEMPERATURES AND RELATIVE HUMIDITIES

ITEM	SMALL EAHP	LARGE EAHP
1	35°C DB, 23.9°C WB 40% RH	35°C DB, 23.9°C WB 40% RH
2	30.5°C, 50% RH	30°C, 53% RH
3	20°C, 88% RH	17.7°C, 90% RH
4	20°C, 88% RH	17.7°C, 90% RH
5	24.4°C, 66% RH	22.9°C, 69% RH
6	41.6°C, 23% RH	46.6°C, 19% RH
MOISTURE REMOVED LITRE/DAY	9.0	13.3

FIG. 12 EXHAUST AIR HEAT PUMP EQUIPMENT ARRANGEMENT FOR USE AS A BASEMENT DEHUMIDIFIER

TABLE 1

ACTUAL HOUSE VENTILATION RATE (I) WITH A BALANCED VENTILATION SYSTEM FOR A HOUSE WITH A FORCED VENTILATION RATE (V_v) OF 0.5 ACH AND A NATURAL INFILTRATION RATE OF (I_o) OF 0.2 ACH

Δt (°C)	R*	I (ACH)
10	0.41	0.41
20	0.58	0.49
30	0.71	0.56
40	0.82	0.61
50	0.92	0.66
60	1.00	0.70

*Correlation for calculating R is based on data for $20°C < \Delta t < 50°C$

TABLE 2

COMPARISON OF LARGE AND SMALL EAHP FOR SPACE HEATING

		Large	Small
Evaporator:	Nominal tube size (mm)	8	8
	No. of rows	3	2
	No. of fins per metre	470	470
	Frontal area (m^2)	0.068	0.056
	Airflow rate (std L/s)	70.8	70.8
	Fan power required (W)	50	50
Condenser:	Nominal tube size (mm)	8	8
	No. of rows	3	2
	No. of fins per metre	470	470
	Face area (m^2)	0.046	0.035
	Airflow rate (std L/s)	70.8	70.8
	Fan power required (W)	50	50
Compressor:	Weight (kg)	20	12.6
	Power supply required	1 ph/240V	1 ph/120V
	Locked rotor current (A)	47.0	32.7
	Power required at 0°C suction and 55°C condensing temperature (W)	775	560
	Refrigerant type	R-12	R-12
Heat Pipe:	Effectiveness (%)	30	30
Electric heater capacity (kW)		1.0	0.75 & 1.5
Supply air temperature (°C)		21.6	21.6
Supply air relative humidity (%)		40	40
Exhaust air temperature (°C)		-25 to 15	-25 to 15
Exhaust air relative humidity (%)		50 to 100	50 to 100
Minimum air temperature from evaporator (°C)		0.56	1.67
Total EAHP capacity (kW)*		3.01	2.47
Total system COP*		5.06	6.18

* Values quoted are for steady state conditions at 21.6°C, 40% RH indoor and -25°C outdoor temperatures, and do not include any input from the electric heaters.

TABLE 3

FAN REQUIREMENTS

Fan	Airflow rate (std L/s)	Pressure loss (Pa)	Power req'd by fan (watts)
Furnace fan	-	-	250
Supply Fan	70.8	40	30
Exhaust Fan	70.8	40	30
ATAHE-delivery	70.8	65	45
-exhaust	70.8	65	45
Small EAHP			
- delivery (condenser)	70.8	75	50
- exhaust (evaporator)	70.8	75	50
Large EAHP			
- delivery (condenser)	70.8	75	50
- exhaust (evaporator)	70.8	75	50

TABLE 4

ENERGY COSTS IN 1985 DOLLARS

	Vancouver B.C.		Saskatoon Sask.		Montreal Que.	
	Gas[1]	Elect[2]	Gas	Elect	Gas	Elect
Energy Cost ($/GJ)	7.82	14.00	6.79	14.04	10.43	11.36

1. Gas prices are for June 1985 and for a monthly consumption of $283 m^3$. Energy value of gas is 37.259 MJ/m^3. Average furnace efficiency of 70% is assumed.

2. Electricity prices are based on the average bill for a consumption of 750 kW-hrs/month as of June 1985.

3. Information supplied by Energy Mines & Resources, Energy Statistics Handbook.

TABLE 5

COMPARISON OF ENERGY REQUIREMENTS FOR VENTILATION DEVICES

Location & recovery device used	Total heat req'd (GJ)	On time EAHP Compressor (hrs)	On time of furnace (hrs)	Energy recov'd from exhaust (GJ)	Energy del'd by ventilation device (GJ)	Elect. energy to ventilation device (GJ)	Elect. energy for furnace fan (GJ)
MONTREAL, Que.							
Base case	55.47	-	1484	-	0.71	1.42	1.34
ATAHE		-	883	21.41	22.47	2.12	0.80
Small EAHP		4,819	491	27.28	37.12	11.02	0.44
Small EAHP c/w Heaters		4,819	317	26.97	43.64	17.91	0.29
Large EAHP		4,402	314	31.87	43.72	13.03	0.28
Large EAHP c/w Heater		4,402	232	31.52	46.79	16.43	0.21
SASKATOON, Sask.							
Base case	77.39	-	2078	-	0.71	1.42	1.87
ATAHE		-	1300	27.88	28.94	2.12	1.17
Small EAHP		5382	924	32.37	42.95	11.75	0.83
Small EAHP c/w Heaters		5382	598	32.17	55.12	24.13	0.54
Large EAHP		5090	696	38.36	51.44	14.26	0.63
Large EAHP c/w Heaters		5090	515	37.98	58.19	21.39	0.47
VANCOUVER, B.C.							
Base case	31.27	-	828	-	0.71	1.42	0.75
ATAHE			398	15.08	16.26	2.12	0.36
Small EAHP		4221	36	20.82	29.92	10.28	0.03
Small EAHP c/w Heaters		4221	29	20.80	30.16	10.54	0.03
Large EAHP		3402	-	21.97	31.69	10.92	0
Large EAHP c/w Heaters		3402	-	21.97	31.76	10.98	0

Note: The exhaust & supply Fans for each ventilation device will operate continuously for the 6552 hour heating season.

TABLE 6

COMPARISON OF ENERGY COSTS ASSOCIATED WITH VENTILATION DEVICE

Location and recovery device used	Gas Furnace			Electric Furnace		
	Cost of heating & ventilation (1985$)	Energy cost savings of recovery compared to base case (1985$)	Simple pay back period for recovery device (yrs)	Cost of heating & ventilation (1985$)	Energy cost savings of recovery compared to base case (1985$)	Simple pay back period for recovery device (yrs)
MONTREAL, Que.						
Base case	604.00	-	-	653.00	-	-
ATAHE	377.00	225.00	2.6	408.00	245.00	2.4
Small EAHP	322.00	281.00	2.8	339.00	315.00	2.5
Small EAHP c/w Heaters	330.00	272.00	2.9	341.00	312.00	2.8
Large EAHP	274 00	329.00	2.8	285.00	369.00	2.5
Large EAHP c/w Heater	280.00	323.00	3.2	288.00	366.00	2.8
SASKATOON, Sask.						
Base case	567.00	-	-	1,123,00	-	-
ATAHE	375.00	192.00	3.0	726.00	396.00	1.5
Small EAHP	410.00	156.00	5.0	660.00	463.00	1.7
Small EAHP c/w heaters	498.00	69.00	12.7	659.00	464.00	1.9
Large EAHP	385.00	182.00	5.1	573.00	549.00	1.7
Large EAHP c/w Heaters	437.00	130.00	7.9	576.00	546.00	1.9
VANCOUVER, B.C.						
Base case	269.00	-	-	458.00	-	-
ATAHE	152.00	117.00	5.0	245.00	213.00	2.7
Small EAHP	155.00	114.00	6.8	163.00	295.00	3.0
Small EAHP c/w Heaters	148.00	121.00	7.3	163.00	295.00	3.09
Large EAHP	151.00	119.00	7.9	151.00	307.00	3.0
Large EAHP c/w Heaters	151.00	118.00	8.7	151.00	308.00	3.4

TABLE 7

ESTIMATED COST OF VENTILATION DEVICES

DEVICE	CAPITAL COST ($1985)	INSTALLATION COST ($1985)	TOTAL COST ($1985)
Base E&SF	120	700	820
ATAHE	700	700	1400
SMALL EAHP	900	700	1600
SMALL EAHP c/w Heaters	950	750	1700
LARGE EAHP	1050	700	1750
LARGE EAHP c/w Heaters	1100	750	1850

TABLE 8

HEATING PROVIDED BY EXHAUST AIR HEAT PUMP FOR HEATING SEASON

LOCATION & EAHP USED	HEATING REQUIRED BY HOUSE (GJ)	USEFUL HEAT SUPPLY BY EAHP (GJ)	PERCENTAGE OF HOUSE HEATING LOAD MET BY EAHP (%)
MONTREAL, Que.			
Small EAHP	55.47	37.12	66.9
Small EAHP c/w Heaters		43.64	78.7
Large EAHP		43.72	78.8
Large EAHP c/w Heaters		46.79	84.4
SASKATOON, SASK.			
Small EAHP	77.39	42.95	55.5
Small EAHP c/w Heaters		55.12	71.2
Large EAHP		51.44	66.5
Large EAHP c/w Heater		58.19	75.2
VANCOUVER, B.C.			
Small EAHP	31.27	29.92	95.7
Small EAHP c/w Heaters		30.16	96.5
Large EAHP		31.69	100
Large EAHP c/w Heater		31.76	100

APPENDIX A

CHARACTERISTICS OF HOUSE MODEL, WEATHER DATA, AND HOUSE HEATING REQUIRED FOR BASE CASE AND ATAHE.

Characteristics of House Model

The standard house chosen for comparison of various ventilation practices is meant to represent a typical detached house with heated basement which has been constructed to conform to R-2000 type specifications. The insulation values, size of south facing windows etc. were taken from the average figures given in a monitoring report of constructed R-2000 houses across Canada (Dumond, 1984). Two adults and two children are assumed to occupy the house. The following references were used as a guide when determining the heat loss of the house (Cane, 1979 and Dumont, et al., 1982). Details are given below:

Mechanical Ventilation for winter conditions (V_v)	0.5 air changes/hour
Air infiltration for winter conditions (I_o)	0.2 air changes/hour
Living Area	106 m^2 (1,141 ft^2)
Basement Area	106 m^2 (1,141 ft^2)
Total house volume, for 2.44 m (8 ft) ceiling	517 m^3 (18,267 ft^3)
Thermal conductance of building skin above grade	65.3 W/°C (123.5 Btu/hr-°F)
Below grade heat loss (assumed constant)	0.4 kW (1,365 Btu/hr)
Gain from appliances, lights, people and sun (assumed constant)	1.55 kW (5,292 Btu/hr)
Loss due to a total ventilation rate of 0.7 ACH (V_v+I_o)	179.7 W/°C (340 Btu/hr-°F)

The house heating required for 0.4 ACH and 0.7 ACH is calculated in Table A1 for the various temperature increments (bins). Table A2 lists the total seasonal heating requirement for the base house (0.7 ACH) and Table A3 lists the total seasonal heating requirement for the ATAHE house (0.4 ACH) for each location.

CANE, R.L.D. September 1979. A "Modified Bin Method for Estimating Annual Heating Requirements of Air Source Heat Pumps. ASHRAE Journal: 60-63.

DUMONT, R. December 1984. R-2000 Home Monitoring Report, Interim Monitoring Results. Division of Building Research, Saskatoon, National Research Council Canada.

DUMONT, R., M.E. LUX AND H.W. ORR. September 1982. R-2000 Energy Analysis Program, Rev. 4.02.2, based on NRC Hotcan 2.0. DBR Computer Program No. 49, Division of Building Research, National Research Council Canada.

TABLE A1

Heat Loss/Gain for Base House & ATAHE House

Outdoor Temp. Range (°C)	Heat Loss with ATAHE (.4ACH) (kW)	Base House Heat Loss (.7ACH) (kW)	Below Grade Heat Loss (kW)	Heat Gain (kW)	House Heating Req'd (.4ACH) (kW)	House Heating Req'd (.7ACH) (kW)
18.3 - 20.5	0.22	0.34			0	0
15.5 - 14.9	0.58	0.80			0	0
12.7 - 14.9	0.94	1.30			0	0.15
9.9 - 12.1	1.30	1.80			0.15	0.65
7.1 - 9.3	1.67	2.30			0.52	1.15
4.3 - 6.5	2.03	2.80			0.88	1.65
1.5 - 3.7	2.40	3.30			1.25	2.15
-1.3 - 0.9	2.76	3.80	0.4	1.55	1.61	2.65
-4.1 - -1.9	3.12	4.30			1.97	3.15
-6.9 - -4.7	3.49	4.80			2.34	3.65
-9.7 - -7.5	3.85	5.30			2.70	4.15
-12.5 - -10.3	4.21	5.80			3.06	4.65
-15.3 - -13.1	4.58	6.30			3.43	5.15
-18.1 - -15.9	4.94	6.80			3.79	5.65
-20.9 - -18.7	5.30	7.30			4.15	6.15
-23.7 - -21.5	5.67	7.80			4.52	6.65
-26.5 - -24.3	6.04	8.30			4.89	7.15
-29.3 - -27.1	6.40	8.80			5.25	7.65
-32.1 - -29.9	6.76	9.30			5.61	8.15
-34.9 - -32.7	7.13	9.80			5.98	8.65
-37.7 - -35.5	7.49	10.30			6.34	9.15
-40.5 - -38.3	7.85	10.80			6.70	9.65
-43.3 - -41.1	8.22	11.30			7.07	10.15

TABLE A2

Total Seasonal Heating Requirements for Base House (.7 ACH)

Temp. Range (°C)	Heating Req'd (kW)	Vancouver B.C.		Montreal Que.		Saskatoon Sask.	
		hrs	GJ	hrs	GJ	hrs	GJ
18.3 - 20.5	0	105	0	227	0	126	0
15.5 - 17.7	0	289	0	329	0	192	0
12.7 - 14.9	0.15	642	0.35	432	0.23	287	0.15
9.9 - 12.1	0.65	1042	2.44	504	1.18	382	0.89
7.1 - 9.3	1.15	1505	6.23	519	2.15	429	1.78
4.3 - 6.5	1.65	1398	8.30	588	3.49	468	2.78
1.5 - 3.7	2.15	852	6.59	717	5.55	518	4.01
-1.3 - 0.9	2.65	457	4.36	674	6.43	603	5.65
-4.1 - -1.9	3.15	150	1.70	532	6.03	527	5.98
-6.9 - -4.7	3.65	50	0.66	452	5.94	481	6.32
-9.7 - -7.5	4.15	24	0.36	391	5.84	416	6.22
-12.5 - -10.3	4.65	10	0.17	325	5.44	355	5.94
-15.3 - -13.1	5.15	4	0.07	262	4.86	325	6.03
-18.1 - -15.9	5.65	2	0.04	171	3.48	283	5.76
-20.9 - -18.7	6.15			111	2.46	272	6.02
-23.7 - -21.5	6.65			63	1.51	233	5.58
-26.5 - -24.3	7.15			22	0.57	195	5.02
-29.3 - -27.1	7.65			7	0.19	147	4.05
-32.1 - -29.9	8.15			3	0.09	94	2.76
-34.9 - -32.7	8.65			.5	0.02	44	1.37
-37.7 - -35.5	9.15			.5	0.02	20	0.66
-40.5 - -38.3	9.65					8	0.28
-43.3 - -41.1	10.15					1	0.04
Totals		6528	31.27	6329	55.47	6401	77.39

TABLE A3

Total Seasonal Heating Requirements for ATAHE House (.4 ACH)

Temp. Range (°C)	Heating Req'd (kW)	Vancouver B.C.		Montreal Que.		Saskatoon Sask.	
		hrs	GJ	hrs	GJ	hrs	GJ
18.3 - 20.5	0	105	0	227	0	126	0
15.5 - 17.7	0	289	0	329	0	192	0
12.7 - 14.9	0	642	0	432	0	287	0
9.9 - 12.1	0.15	1042	0.56	504	0.27	382	0.21
7.1 - 9.3	0.52	1505	2.82	519	0.97	429	0.80
4.3 - 6.5	0.88	1398	4.43	588	1.86	468	1.48
1.5 - 3.7	1.25	852	3.83	717	3.23	518	2.33
-1.3 - 0.9	1.61	457	2.65	674	3.91	603	3.49
-4.1 - -1.9	1.97	150	1.06	532	3.77	527	3.74
-6.9 - -4.7	2.34	50	0.42	452	3.81	481	4.05
-9.7 - -7.5	2.70	24	0.23	391	3.80	416	4.04
-12.5--10.3	3.06	10	0.11	325	3.58	355	3.91
-15.3--13.1	3.43	4	0.05	262	3.24	325	4.01
-18.1--15.9	3.79	2	0.03	171	2.33	283	3.86
-20.9--18.7	4.15			111	1.66	272	4.06
-23.7--21.5	4.52			63	1.03	233	3.79
-26.5--24.3	4.89			22	0.39	195	3.43
-29.3--27.1	5.25			7	0.13	147	2.78
-32.1--29.9	5.61			3	0.06	94	1.90
-34.9--32.7	5.98			.5	0.01	44	0.95
-37.7--35.5	6.34			.5	0.01	20	0.46
-40.5--38.3	6.70					8	0.46
-43.3--41.1	7.07					1	0.03
Totals		6528	16.19	6329	34.06	6401	49.51

APPENDIX B

DESCRIPTION OF CALCULATIONS MADE FOR COMPARISON OF VENTILATION DEVICES

B1.0 BASE CASE

For the base case it was assumed that the house described in Appendix A was ventilated using an exhaust and supply fan (E&SF) requiring 30 watts of power each. These fans were run continuously for the 6552 hours of the heating season and thus required a total of 1.42 GJ of electrical energy to maintain the house ventilation rate at 0.5 ACH. Heating was accomplished by either an electric or gas furnace of similar capacity. Typically, gas furnaces come in discrete sizes of 4.4 kW (15,000 Btu/hr) with a minimum capacity of 11.7 kW (40,000 Btu/hr). Electric furnaces come with 5 kW (17,000 Btu/hr) increments in size. From Table A2 in Appendix A, the heating requirements for the house in each location were calculated. Furnace size was selected at 10.0 kW, which provided sufficient heating capacity in all locations. It was assumed that a 250 watt furnace fan is required and that the fan was "fully loaded". The assumption was also made that all the electrical power to the furnace fan, P_{ff}, and the electrical power consumed by the ventilation supply fan, P_{sf}, ended up as heat to the house.

The operating hours for the furnace, Δt_f, were calculated by:

$$\Delta t_f = \frac{(Q_h - P_{sf})}{(Q_f + P_{ff})} \qquad (B1)$$

where Q_h is the total energy (GJ) required to heat the house and Q_f is the heating capacity of the furnace (kW). Note that in this and subsequent equations the conversion of units to make the values compatible has not been explicitly detailed for the sake of readability.

The cost of heating and ventilation for the base case was calculated by:

$$D_B = (Q_h - (P_{sf} + P_{ff} \times \Delta t_f)) \times C_f + (P_{ff} \times \Delta t_f + 2 \times P_{sf}) \times C_e \quad (B2)$$

where: C_f = cost of energy for the furnace (1985 dollars per GJ delivered)

C_e = cost of electrical energy (1985 dollars per GJ).

B2.0 AIR-TO-AIR HEAT EXCHANGER (ATAHE)

It was assumed that the same furnace would be used to heat the house as in the base case. The ATAHE will operate continuously for the heating season; therefore the two 45 watt fans will consume a total of 2.12 GJ of electrical energy. The heat recovered by the ATAHE, will be 60% of the heat lost by the exhaust air (0.5 ACH). The total energy required to heat the house with the ATAHE installed, Q_{ha}, is listed in Table A3 in Appendix A for all three locations. Note that heat losses for ambient temperatures greater than 15°C were neglected because the house does not require heating at these temperatures.

Since the ATAHE will reduce the heating load of the house, the furnace will have to supply less energy and will operate for a shorter time.

$$\therefore \quad \Delta t_f = \frac{(Q_{ha} - P_{sf})}{(Q_f + P_{ff})} \quad (B3)$$

Where: P_{sf} = Energy consumed by ATAHE supply fan (GJ)

Therefore, the total cost of heating and ventilating the house with an ATAHE and regular furnace is given by:

$$D_{HE} = (Q_{ha} - (P_{sf} + P_{ff} \times \Delta t_f)) \times C_f + (P_{ff} \times \Delta t_f + 2 \times P_{sf}) \times C_e \quad (B4)$$

B3.0 EXHAUST AIR HEAT RECOVERY HEAT PUMP (EAHP)

The EAHP will operate from the first stage of the house thermostat and the furnace (same furnace as in the base case) will be operated on the second stage of the thermostat. Once again, the electrical power to the supply fan, P_{sf}, will be considered as a contribution to space heating. The average capacity, Q_d, of the EAHP is given by:

$$Q_d = Q_c + P_{sf} \tag{B5}$$

where Q_c is the steady-state capacity of the EAHP,(this includes the heat pump plus heat pipe and electric heaters when operating).

The house will not need heating when ambient temperatures are greater than 15.0°C as the heat loss by the house will be less than the internal heat gains. The EAHP will cycle on and off with the thermostat for ambient temperatures between 15.0°C and the balance point.

The balance point is defined as the ambient temperature at which the heat loss from the house is just "balanced" by the output of the heat pump. For the small EAHP the balance point was 3.6°C and for the large EAHP it was 0.0°C. For each temperature bin, j, the following relations must be satisfied:

$$Q_d \times \Delta t_{on} < Q_j \times \Delta t_j \tag{B6}$$

$$\Delta t_j = \Delta t_{on} + \Delta t_{off} \tag{B7}$$

where: Q_j = the average rate of heat lost from the house for temperature bin j (kW)
Δt_j = number of hours in temperature bin j
Δt_{on} = 'on' time of the EAHP (hrs)
Δt_{off} = 'off' time of the EAHP, i.e. accumulated time (hrs) that the first stage of the house thermostat was not activated

The time during which the furnace will operate is given by:

$$\Delta t_f = \frac{Q_h - Q_d \times \Delta t_{on}}{(Q_f + P_{ff})} \tag{B8}$$

Therefore, the total cost of heating and ventilating the house with an EAHP and regular furnace is given by:

$$D_{HP} = (Q_h - (Q_d \times \Delta t_{on} + P_{ff} \times \Delta t_f)) \times C_f + (P_{ff} \times \Delta t_f + Q_{vf} + P_c \times \Delta t_{on}) \times C_e \quad (B9)$$

where $Q_{vf} = (P_{sf} + P_{ef}) \times \Delta t_s$ (B10)

P_{ef} = power required by exhaust fan

Δt_s = number of hours in the heating season (6552).

P_c = Power required by compressor and electric heaters

B4.0 FANS

Each device will have two fans. The power to operate these fans will be a direct charge against the cost of operating the device. For these small, inefficient fans and motors an overall electricity to air efficiency, E_F of 10% will be assumed. The power required by each fan (watts) is calculated using the equation B11; where Δp is the pressure differential supplied by the fan to move the air (P_a) and V_v is the airflow rate (L/s).

$$P_F = \frac{\Delta p \; V_v}{E_F} \times \frac{1}{1000} \quad (B11)$$

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NOTES

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14 Energy Recovery; Ventilation; Infiltration; Air Quality; Heat Pump; Heat Pipe; Air-to-Air Heat Exchanger

SUMMARY/SOMMAIRE

15 An Exhaust and Supply Fan, Air-to-Air Heat Exchanger and Exhaust Air Heat Pump with Heat Pipe incorporated are compared to determine the most economic device for forced ventilation requirements in R-2000 houses.

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